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TRABAJO TEORICO-EXPERIMENTAL

An experimental study of heat transfer enhancement using vortex generators in a finned elliptical tube

Estudio experimental de la intensificación de la transferencia de calor en un tubo elíptico aletado

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Abstract/ Resumen
In the recent years some papers have dealt with heat exchangers with non-circular tube geometry and vortex generators. Parametric studies with vortex generators applied on heat exchangers models are necessary. The present work consists of an experimental investigation about the influence of combining a plate-finned elliptical tube with vortex generators mounted on the fin surface. The objective is to search for the best position to place the vortex generators just considering heat transfer enhancement. The experimental technique for Nussel tnumber determination was the naphthalene sublimation and mass transfer analogy. A visualization was implemented spreading wetted chalk on fin surface, as a color sensitive evaporation material. The better position obtained is useful for future investigations involving heat exchanger formed by a higher number of tube rows. Additionally was studied the influence of Reynolds number and angle of attack on the average Nusselt number.
Keywords: vortex generators, compact heat exchanger, elliptical tube, naphthalene.

En los últimos años la temática de intercambiadores de calor con tubos de geometría no circulares y generadores de vórtices ha sido investigada, no obstante aún es necesario realizar estudios paramétricos de estos elementos combinados. El trabajo consiste en la investigación de la combinación de generadores de vórtices colocados en la superficie de las aletas y tubos elípticos. El objetivo es buscar la mejor posición para la colocación de los generadores de vórtices, considerando solo el efecto de intensificación de la transferencia de calor. Se utilizó la técnica experimental de sublimación de naftaleno. Se implementó una técnica de visualización basada en la coloración de yeso como función de su contenido de agua. La mejor posición fue obtenida, y es útil para futuros estudios con intercambiadores de calor con un mayor número de filas de tubos. Se estudió además la influencia del número de Reynolds y el ángulo de ataque del generador de vórtices sobre el número de Nusselts.
***Palabras clave**: generadores de vórtices, intercambiadores de calor compactos, tubos elípticos, sublimación de naftaleno.*

INTRODUCTION

Heat transfer enhancement in compact heat exchangers is a recurrent subject in theliterature. Different types of vortex generators have been used with this purpose on thefins surface of compact heat exchanger and other geometries, like ribbed channels, looking to improve its thermal performance [1]. Not only in heat exchangers but also in other applications like conduits, the vortex generatorshave demonstrated its influence on the enhancement of heat transfer[2]. Some applications of vortex generators point to achieve a drag reduction in aerodynamic applications[3]. The generated vortex system produces locally a velocity increasing, thinning of boundary layer and mixing of fluids with different temperatures. The mechanisms mentioned above are responsible for the increasing of local and average heat transfer coefficients noted by different authors in the last years. Flow losses reduction, in some cases, especially for high Reynolds numbers have been observed too. The complex relationships between the vortex system and the main flow in a heat exchanger passage are very difficult to predict, therefore, is necessary to study separately each application. Fore fixed heat exchanger geometry, the vortex generator parameters, e.g. position, angle of attack, aspect ratio and generator's shape must also be investigated.

Ximenes*apud* Pérez (2012), carried out a study of heat exchangers with fined elliptical tubes using of naphthalene sublimation technique [4]. Heat exchangers models with transversal andlongitudinal dimensionless pitches of 3.25 and 2.56 respectively were tested. The average and local Nusselt numbers for models with one and two rows were obtained experimentally. TheReynolds number was varied between 200 and 1700. Additionally three values for the tubes eccentricity, 0.5; 0.65 and 1, were studied. There were not found marked differences in heat transfer between circular and elliptical tubes, independently of the value of the eccentricity. The explanation for the above conclusion was found in the balance between a smaller wake region and a less significant horseshoe vortex system, when the eccentricity is diminished from 1, for circular tubes, to 0.5 for elliptical tubes. The pressure drop wasn't measured;however the paper has a relevant importance because it constitutes an initial reference to studies of heat exchangers with finned elliptical tubes.
Fiebig*et al*(1994) using liquid crystal thermography provided friction and heat transfer data for heat exchangers with three rows of staggered flat and round tubes for smooth fin and using vortex generators [5]. The Reynolds number, height channel based, was varied between 600 and 3000. As a result of comparing round and flat tubes the influence of vortex generators was determinant in heat exchangers elements with flat tubes. The Nusslenumber ratio for enhanced smooth surface was several times higher for flat tubes as comparedto round tubes configuration. The smooth inline arrangement, with a natural poor thermal performance, is the most promissory for using vortex generators, because of longer path traveling vortex. The pressure drop, dramatically increased for flat tube with vortex generators, was however half of the same parameter for enhanced round tubes configuration.

Rocha *et al*(1997), developed a numerical two dimensional model to compare elliptical and round tubes. The results of this work have shown a fin efficiency gain of up to 18%as compared with circular[6]. In addition the higher efficiency values were observed in tubes with eccentricity of 0.5.Bordalo*and*Saboya (1999), carried out a pressure drop experimental study for ellipticaltubeusing the same eccentricity values [7] than Ximenes*apud* Perez (2012). The aerodynamic advantage of elliptical tubes, as compared to round, was found for Reynolds number based in channel height exceeding 1000. A viscous drag higher than form drag, present at low velocity, was responsible for this behavior.

Tiwari et al (2003), carried out a numerical study to determine the flow structure and heat transfer in a rectangular channel with a built-in oval tube and delta winglet type vortex generators in various configurations[8]. The Reynolds number, based on the channel height, was 1000. The results indicated that a certain configuration (4 winglet pairs arranged around the oval tube) produced a mean span-averaged Nusselt number 100% higher when comparedto the plain fin case. All configurations studied confirmed that this type of vortex generatorscan be used to enhance the heat transfer in this type of fin-tube configuration.

Henze*et al*. (2011), using an experimental approach, investigated extensively the flow and heat transfer characteristics behind vortex generators[9]. These vortex generators were solid tetrahedral bodies. A thermocromic liquid crystal was used for heat transfer measurement and a particleimagevelocimetry system allowing them to study the flow field. The effect of the longitudinal vortices induced by the vortex generators was determined. A primary and secondary vortex system were found responsible for a marked enhancement heat transfer, beginning at the trailing edge of the vortex generators. A common flow down area between vortexes was used. The experiments were conducted at higher Reynolds numbers and with the objective of becoming a

databank as a reference for numerical research. The erroranalysis is presented with 95% of confidence. The full databank was placed in a website andis available in text format.
A comprehensive review about the vortex generators and fin tube in heat exchangers was present by Biswas*et al*. (2012). They summarize the current state of the art of this technology [10]. The main trend of both numerical and experimental approaches to investigate the enhancement of heat transfer and pressure drop, as result of vortex generators located or punched on fin surface, are presented. The paper begins considering the influence of the vortex generators and how the pioneer works of Fiebig, Yanagihara and Torri were unraveling the mechanism involved during the complex flow interactions inside a channel withprotrusions which are responsible for the generated vortex. Finally they concluded how the analysis techniques used today for investigating this subject can be used with ample confidence.

In Kattea (2012) an experimental studied of square and circular vortex generators as turbulence promoters in front of heat exchangers is made [11]. Heat transfer enhancement of up to of 39%was achieved when compared to the same heat exchanger but without vortex generators.The distance from the turbulence promoters to the first row was investigated too. Theflow was in turbulent regime and high Reynolds numbers were used (62000-125000). Noinformation is presented about the pressure drop but considering the configuration used itmust be considerably large.

He et al. (2012), investigated numerically arrays of delta-winglet pairs with two layout modes of continuous and discontinuous winglets punched on the fins of heat exchanger with two rows of round tubes in line configuration. The heat transfer performance of two array ofvortex generators arrangements were compared to a conventional large winglet configurationfor the Reynolds number ranging from 600 to 2600 based on the tube collar diameter [12].A common flow up between each pairs of vortex generators was employed. The angle of attack of delta winglets was varied. Arrays with discontinuous winglets show the best thermal performance with a significant augmentation of up to 33.8-70.6% and a pressure drop augmentation of 43.4-97.2% for angle of 30^{o} when compared to the smooth fin. The interactionbetween the different vortexes generated by each vortex generator is an important fact that must be considered because the interaction, in some cases, produce weakening of vortexwith the corresponding shorter length of influence. Additionally a more important role ofthe corner vortex compared to the main vortex is enunciated when the vortex generators are punched on the fin surface. For the analysis two performance evaluation criteria of area goodness and volume goodness were employed.

The heat transfer enhancement using vortex generators was extended to a solar air heater by Bekele*et al*. (2013). The work has both numerical and experimental approaching and an important heat transfer enhancement was found [13]. Nusselt number for the enhancement surface was 3.6 times the same parameter for the smooth surface. This paper investigate the geometry of the vortex generators and its distribution on the surface. The best results were found for vortex generators having height relative to the height channel of 0.5 and a relative longitudinal pitch of about 1.5. A thermo hydraulic performance parameter recommended by Eckert was used for comparison purposes. Reynolds numbers are within the turbulent regime. The experiment was conducted at indoor conditions to guarantee reproducibility in contrast with the lack of reproducibility of solar radiation.

A next generation of works which investigate combined enhancement techniques have begun to appear in the literature Zhang *et al*. (2012) [14] or Huisseune*et al*(2013)where louvered fins were combined with vortex generators[15]. Chompookham*et al* (2010) e.g. investigated experimentally the combination of delta winglet pair with wedge ribbed in turbulent regime and several combination of both structures were tested [1].The channel in smooth condition was used for validation of the experimental technique by comparison with the Dittus-Boelter equation and they match almost perfectly. When the effect of rib arrangement is analyzed the friction factor of staggered configuration of wedge with ribs pointing upstream has a lower value than the in-line and rib pointing downstream arrangement. All configurations implemented in the work achieve an increase in the heat transferred but with an important drop in the pressure.
Zhang *et al*. (2012), developed an experimental study of double pipe heat exchanger but combining helical fins with vortex generators of several types[14]. The Nusselt number and the friction factor were investigated and was found a better performance of the heat exchanger using the combined enhancement when compared to the heat exchangers only using helical fin. The delta winglet vortex generators configured in pairs are more promising than the rest of vortex generators

configuration. The vortex generators were located on the tube, not on the fins as usually happens in compact heat exchangers. The destabilization of fluid flow and a higher turbulence, combined with a higher heat transfer area were identified as mechanism of enhancement heat transfer; however the authors need to clarify the enhancement mechanism studying the flow field.
Huisseune *et al*. (2013), used a numerical approach to investigate the enhancement in compound design consisting of vortex generators punched in heat exchangers with three rows of round tubes and louvered fins in laminar regime[15]. They confirmed three mechanisms for heat transfer enhancement, a thinner boundary layer, a separated flow region delayed and the penetration and mixed of main flow in the wake area. To determinate numerically the point of separation, studies were made to the value of the *x* component of the wall shear stress. The separated flow delayed on the tube surface reduce the drag but, at the same time, a higher frontal area produces an increasing of form drag. The authors enunciated a higher pressure drop as result of this combined effects. They found higher pressure dropsand a minor heat transfer area for the louvered fins heat exchanger with vortex generatorscompared to the baseline design without vortex generators. The comparison shows how for the same heat transferred and pumping power, a more compact heat exchanger is possible.The authors considered only one geometry.

Although in the recent years some papers have dealt with heat exchangers with noncircular tube geometry and vortex generators, further parametric studies with vortex generators applied on different heat exchangers models are necessary. The present work consists of an experimental investigation about the influence of combining a plate-finned elliptical tube with vortex generators mounted on the fin surface. The objective is to search for the best position to place the vortex generators on the fin surface just considering heat transfer enhancement. The experimental technique used for the determination of the average Nusseltnumber was the naphthalene sublimation because it is reliable, easy to use and with high accuracy. The local visualization was implemented spreading wetted chalk on the fin surface, as a color sensitive evaporation material, in order to get quantitative informationabout the local mass transfer coefficient. The knowledge of the better location is very useful for future investigations on more complex heat exchanger models formed by one, two or higher number of tube rows. Additionally, it also studied the influence of Reynolds numberand angle of attack on the average Nusselt number.

MATERIALS AND METHODS

The experiments were conducted in an instrumented open circuit wind tunnel (fig.1).It consists of a contraction at the inlet, a test section, a diffuser, a centrifugal fan and a discharge tube. A flow straightener and a grid are placed before the test section to guarantee uniformity of the velocity profile at the entrance of the tested model. The test section made of acrylic, with a cross section of 0.26m of width and 0.090m of height, was built in such a way to permit a rapid and easy access to the model. The wind tunnel should be open circuit when naphthalene sublimation technique is used, because it avoids the inner air contamination with naphthalene vapors. The contamination of air could produces errors during the test because it is considered a null naphthalene concentration in the air at inlet of tunnel. A liquid thermometer with resolution of 0.1^{o}C is placed in the rear part of tunnel and is used to measure the temperature of the air flowing through the model. The flow rate is sensed by means of a vortex emission flow meter with 1*%* of uncertainty. The atmospheric conditions are obtained with a barometer of mercury and a hygrometer gave us the air's relative humidity. Finally, an electronic balance of high resolution (10^{-4}g) is used to weigh the test specimen before and after each experiment. A chronometer is in charge of counting time. The finned elliptical tube model was constructed using expanded polyurethane for tubes and acrylic for the fins. Aluminum plate was selected for vortex generators construction.In the figure 1 is presented an image of the smooth model formed by four fins and an elliptical tube located in the center of fins.The figure 2, is showing a half fin substituted by an aluminum fin (labeled) having the same dimension than the others. This special fin has its surface cover with naphthalene because previously it was used like a mold to receive melted naphthalene. The surface of naphthalene emulated the isothermal boundary condition for the fin during a mass transfer experiment. The surface of naphthalene that is going to simulate the fin surface is obtained by casting using a clean glass plate as a cover to produce a perfect smooth surface. Symmetric flow and heat transfer conditions were considered and only a half fin needs to be substituted. Additional details for the experiment technique could be found in the reference Pérez *et al*. [4].

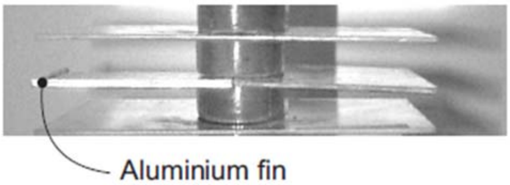

Fig. 1. The smooth finned elliptical model.

The dimensions of model, took from real heat exchangers, used in air conditioning equipment and scaling (1:10) to adequate dimensions for collocation of vortex generators are reflected in dimensionless form in the table 1, with reference to the figure 2.S_L and S_T are the transversal and longitudinal pitches respectively. *l* is the vortex generator thickness.

Table 1. Dimension of model and vortex generators used in this work.

Parameter	Nomenclature	Value	Parameter	Nomenclature	Value
Longitudinal pitch	S_L/D_2	3	Aspect ratio	Λ=2H/b	1
Transversal pitch	S_T/D_2	4	Angle of attack	β	45^o
Fin spacing	E/D_2	0.28	Fin thickness	d/D_2	0.02
VG height	H/D_2	0.28	VG thickness	l/D_2	0.02

In the figure 2, β is the angle of attack, H and b are the height and the base length of vortex generator respectively. Delta shape winglets vortex generators are always used in this work. All dimensionless parameters were obtained with reference to the minor diameter of the ellipse (D_2=6,35*mm*). The eccentricity (D_2/D_1) used was 0.5 because it was found to be the most efficient by Rocha *et al*. [6].

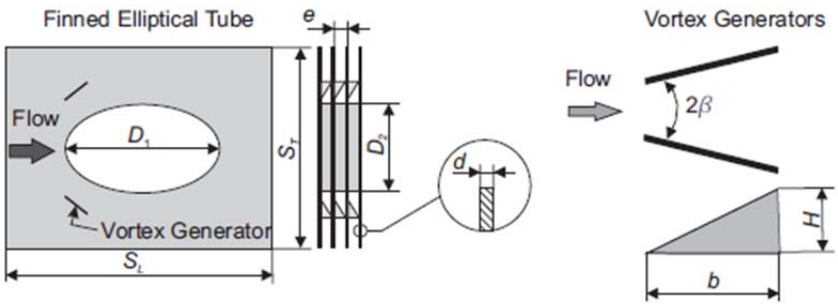

Fig. 2. Parameters of heat exchanger model and vortex generator.

The experimental procedure consisted in assembling the model with the fin covered by naphthalene, previously, the fin have been weighted. The model was carried to the tunnel and placed into the test section. Immediately, the model was removed and the fin was weighted again. This procedure enabled the measurement of the mass of naphthalene lost by natural sublimation (Δm_{ns}) during the manipulation of the model. This value was usedlater for mass correction because only the mass transferred by forced convection during the test inside the tunnel is relevant. The time wasted in the process of place and remove the model from the wind tunnel was assumed constant for every test. Finally, the fan was turn on and a short warm-up period began. This warm-up is necessary to get a stabilization of the temperature in the test room. After the tunnel warm-up, the fin was weighted before to be placed with the model into the test section. The chronometer was immediately started. The air temperature was taken every two minutes and the room conditions were also monitored.After 35 or 40 minutes (Δτ) the fan and the chronometer were turned off and the modelwas removed. The special fin was dismounted and weighted again. As result, an initialand a final weight were available. Additionally the time elapsed and the air temperature are available too. If the room temperature changed more than 0.7°C during the test run, the test was interrupted and the result were not considered. In this work was used a pair of vortex generators having an aspect ratio of equal tone and symmetrically placed in both sides of the tube. A delta winglet vortex generator was chosen and an angle of attack of 45^o was initially used. The vortex generators were always placed in common flow down disposition. The experimental procedure is developed for smooth condition (without vortex generators) and using vortex generators.

Data Reduction Procedure

The average Nusselt number was determined as follows. Each set of measurements was processed beginning with the calculation of the mass transferred by forced convection during the test (Δm), obtained by the difference between the initial and the final weight of the fin ($m_i$andm_f) and considering also the mass lost by natural sublimation Δm_{ns} referredearlier. The mass lost by natural sublimation occurs during transportation of the modelfrom the wind tunnel to the scale and vice versa. The mass transferred by forced convection can be calculated using equation (1):

$$\Delta m = m_f - m_i - \Delta m_{ns} \qquad (1)$$

The averaged mass transfer coefficient h_m was determined by means of equation (2), considering the time elapsed ($\Delta\tau$) during the test:

$$h_m = \frac{\Delta m / \Delta\tau}{\Delta\rho_{log} A_f} \qquad (2)$$

Where A_f is the fin area covered with naphthalene. The mean logarithmic vapor density $\Delta\rho_{log}$was calculated by the following equation (3):

$$\Delta\rho_{log} = \frac{(\rho_{vw} - \rho_{v\infty in}) - (\rho_{vw} - \rho_{v\infty out})}{ln\left(\frac{(\rho_{vw} - \rho_{v\infty in})}{(\rho_{vw} - \rho_{v\infty out})}\right)} \qquad (3)$$

Where the density of vapor at the fin surface level (ρ_{vw}) is calculated using the ideal gas law at the surface temperature. The vapor pressure of naphthalene is obtained by means of Ambrose's correlation. The vapor density of the mainstream at the entrance ($\rho_{v\infty in}$) is considered null because the wind tunnel is open circuit. The vapor density of the mainstream ($\rho_{v\infty out}$) at the exit of the channel was calculated using the equation (4):

$$\rho_{v\infty out} = \frac{\Delta m / \Delta\tau}{Q} \qquad (4)$$

The denominator of equation (4), (Q) is the volumetric air flow rate in the channel, betweentwo consecutive fins. Therefore, the mass transfer Stanton number could be determined bythe following equation (5):

$$St_m = \frac{h_m}{u} \qquad (5)$$

Where u is the average velocity in the minimum free flow area of the channel, function of the flow rate through the tunnel. If the following assumptions are considered: there is not chemical reaction, there is a low rate of mass transfer, the fluid has constant properties and radiation heat transfer is neglected.

Then, from the analogy between the heat and mass transfer we could write as observed in the equation (6) that:

$$St_h = St_m \left(\frac{Pr}{Sc}\right)^{2/3} \qquad (6)$$

whereSt_h is the heat transfer Stanton number and Sc the Schmidt number, obtained fromCho's correlation as function of the surface temperatureT_w. The Schmidt number is obtained from equation (7):

$$Sc = 2.28\left(\frac{T_w}{298.16}\right)^{-0.1526} \qquad (7)$$

The Nusselt number was obtained by the equation (8), where the Reynolds number was calculated using the definition of hydraulic diameter of the channel.

$$Nu = St_h Re Pr \qquad (8)$$

Finally the heat transfer enhancement E was evaluated using an enhancement ratiobetween the averaged Nusselt number for the augmented (Nu) surface and the smooth(Nu_0) surface as indicated by equation (9):

$$E = Nu/Nu_0 \qquad (9)$$

RESULTS AND DISCUSSION

The experimental technique and the procedure used in this work were certified by comparison with experimental results available in the literature. Figure 3, presents the experimental results by Ximenesapud Pérez *et al*.[4], and those obtained in the present work for the same transversal pitch. The values were plotted using the Sherwood number (Sh) to avoid the possible divergence introduced by the exponent of the heat and mass transfer analogy and the Schmidt number. It can be said that the present results are in good similar to previous experiments. Both numbers are based in the hydraulic diameter.

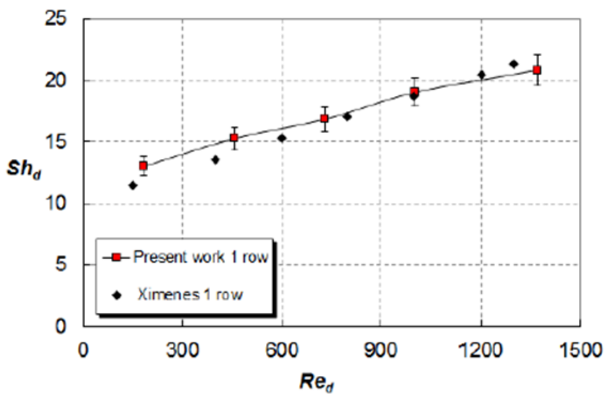

Fig. 3. Sherwood number vs. Reynolds number for Ximenesapud Perez *et al.* [2012] and the present work.

The generators position was varied to sweep the fin surface around the tube. In the abscissas axis, they are plotted numbers from 0 to 16, each one corresponding to a different vortex generators position identified with the same number at figure 4.

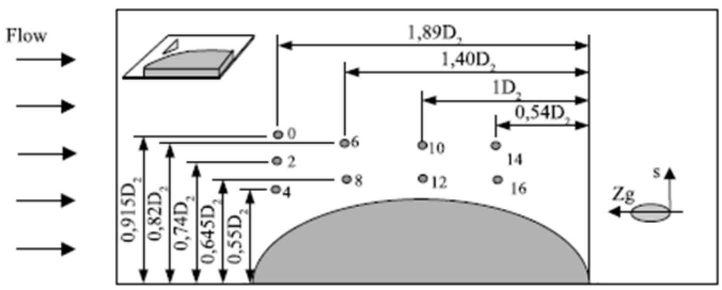

Fig. 4. Vortex generators positions tested.

Figure 5, presents the heat transfer enhancement results for the finned elliptical tube obtained using vortex generators. The initial results plotted in figure 5, show approximately the same behavior for the two Reynolds numbers tested in this work. Generally speaking, the heat transfer enhancement diminishes when the vortex generators is placed farther from the fin leading edge. The region of vortex actuation is increased gradually when the position is dislocated toward the leading edge of the fin. Therefore, the area with a higher local velocity is larger when compared to the undisturbed region. Additionally, another interesting behavior can be noted in the same graphic, which is the increasing of heat transfer when the location is far from the tube surface. The interaction between the vortex generated by delta winglets and the horseshoe vortex generated in front of the tube should be the reason for this fact. In this situation both vortex systems act in the same region (clearer areas in figure 6, obtained using a visualization process, are indicative of higher mass transfer coefficient). When the vortex generators are placed farther from the tube surface, their influence is exerted over a surface area where, in smooth condition, the heat transfer coefficient is lower. In other words, the potential for the heat transfer enhancement is lower when the vortex generator position is closer to the tube surface. The visualization technique used in this work was implemented spreading wetted chalk on a fin surface. The surface with the wetted chalk was exposed to the air flow condition inside the tunnel and the water began to be removed from the chalk. Using a dark background on the fin can be noted clearer areas where the dry process is more intense.

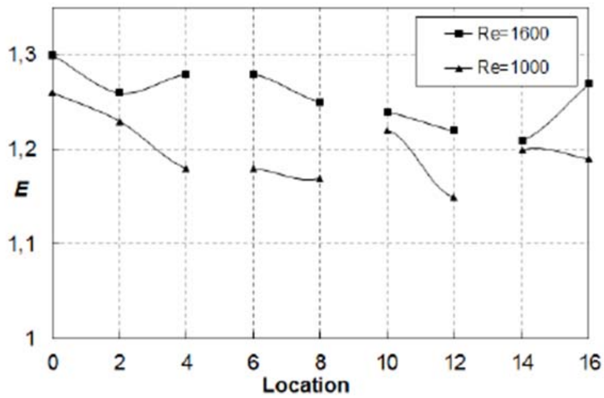

Fig. 5. ***E (Nu=Nu$_0$)*** **as function of the position of vortex generators for two values of Reynolds number.**

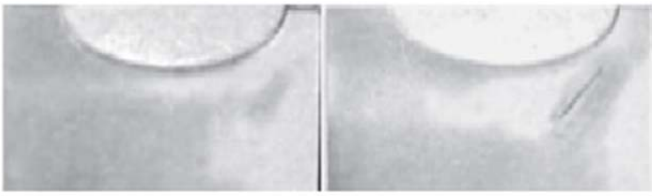

Fig. 6. Pictures showing a common actuation area for both of vortex systems. Left: horseshoe vortex system (smooth fin). Right: both vortex systems (VG at position 4).

An explanation is necessary for the last positions (14 and 16), because their location is near the wake region, where there is no influence of the horseshoe vortices and the heat transfer coefficient is low. The vortex generators in this region deviate the flow to the wake region, producing an increment of the heat transfer in this area. Pestei *et al*. (2005) reported that placing the vortex generators close to the wake region of a round tube improves the thermal performance [16]. Finned elliptical tube with eccentricity of 0.5 presents also are circulation region behind the tube, explaining the results mentioned above. Considering these previous results, another group of experiments were conducted. The area of study was redefined and the vortex generators were placed only around the position where the highest heat transfer enhancement was reached in the preliminary experiments. The new locations intent to redefine the study and they are defined in the figure 7.

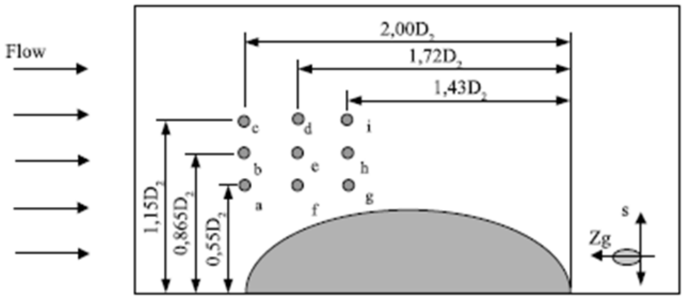

Fig. 7. New locations for vortex generators around the position where the highest heat transfer enhancement was reached.

In the figure 8, the tests for Reynolds numbers 1600 are presented. In this figure, it can be observed the same tendencies already noted in the analysis of the figure 5: the intensification is increased when the VG location is farther from the tube surface and when it is dislocated toward the leading edge of the fin.

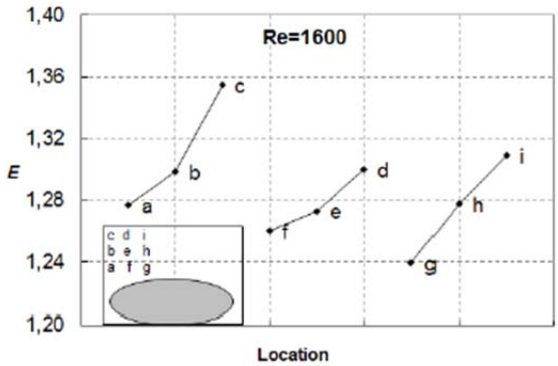

Fig. 8. Values of *E,* (*Nu/Nu_0*), for the new locations around the position where the highest heat transfer enhancement was reached.

The points located at larger distances from the leading edge present the lower values forheat transfer enhancement. However, these values are higher than those for points located at the end of the fin in the preliminary experiments (fig. 4 and 5). The authors did not seekthe locations for the maximum heat transfer enhancement farther from the tubes becauseit has not practical importance in compact heat exchangers as the vortex generator need tobe placed between tubes.The influence of the Reynolds number was analyzed for the point where a highest heattransfer enhancement was found. This plot is shown in the figure 9, where the heat transferenhancement is increased when the Reynolds number is augmented. The ratio Nu/Nu_0 didnot vary linearly with the Reynolds number, probably because of the short length of themodel and the increasing relative importance of the horseshoe vortices system, for highervelocities. The angle of attackof the vortex generator, referred to the main direction of the flow,was varied and five different values were tested for the position “c”. For higher angles of attack the heat transfer increases, reaching a maximum near 55^o.When the angle of attack is further increased, a flow deviation occurs instead of a strongervortex formation.

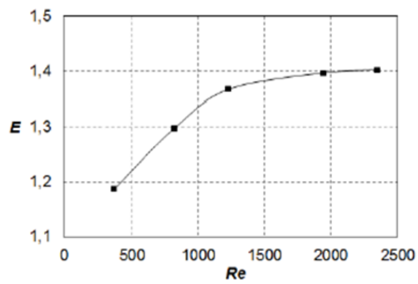

Fig. 9. Influence of the Reynolds number in the heat transfer enhancement.

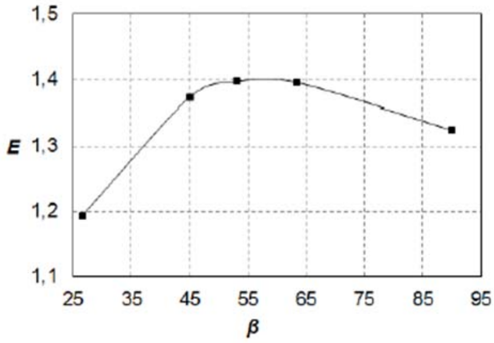

Fig. 10. Influence of angle of attack on the heat transfer enhancement, for Re= 1600.

Every configuration of vortex generators tested in this work has associated a pressure drop. Almost in every case the pressure drop will be higher when compared to smooth surface because of a higher frontal projected area. An incremental in drag must be expected and a combined analysis of pressure drop and heat transfer is necessary to have the whole picture of the problem. In this work it was not studied because the objective was to know the best location considering only the heat transfer. The next step will be to analyze the thermo-hydraulic behavior of more complex geometries having one and two rows of finandelliptical tube.

CONCLUSIONS

Vortex generators enhance the heat transfer when applied to a model of a finned elliptical tube. In the plate-finned elliptical tube, the average mass transfer coefficient was found to be higher when the location of the vortex generator was closer to the leading edge of the fin and farther from the tube surface. The influence on heat transfer of the position of vortex generators on the fin surface was investigated and the best result was obtained for the point's", located at s/D_2 = 2.30 and Zg/D_2 = 2, referred to the tube (figure 7). The heat transfer enhancement rate obtained at this location should not be considered a maximum because it was not detected a diminution beyond it. However, amplification of the range of the study has no practical significance because of tube arrangement constraints. The increasing of the Reynolds number always produced higher heat transfer enhancement for the range of Reynolds number tested. Increasing the angle of attack improved the heat transfer behavior but there is a maximum value for this parameter. In summary, it is possible to improve the thermal performance of a plate-finned elliptical tube by the adequate use of vortex generators. The next step includes studies about the influence of built-in vortex generators on the heat transfer and pressure loss of rows of finned elliptical tubes.

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